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PRESSURE RECOVERY IN A CYLINDRICAL HEAT PIPE AT HIGH RADIAL REYNOLDS NUMBERS AND AT HIGH MACH NUMBERS

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Abstract

The pressure recovery in a cylindrical heat pipe has been investigated. The experiments cover average radial Reynolds numbers between 5 and 150 and average Mach numbers up to the velocity of sound.

During preliminary experiments in a cylindrical, gravity-assisted heat pipe at high Mach numbers large condensate flow instabilities were observed. As a consequence the heat pipe power varied strongly. Based on these observations an improved heat pipe design was made that resulted in steady operating conditions throughout the entire parameter range. This heat pipe is described.

The pressure recovery was measured and compared with results from a two-dimensional analytical model for describing compressible vapor flow in heat pipes. Good agreement with the experimental data was found.

Nomenclature

A	cross-section of vapor duct
c	sonic velocity
c_p	specific heat of coolant
D	diameter of the vapor channel
F	coolant mass flow
f	fanning friction factor
h_{fg}	heat of vaporization
L	length of evaporator or condenser
$\overline{Ma}$	Mach number, averaged over the cross section
Ma_{max}	maximum local Mach number
p	static pressure of vapor
p_0	reference pressure of vapor
Q	heat pipe power
R	universal gas constant
Re	axial Reynolds number
Re_r	radial Reynolds number
$\overline{Re}_r$	radial Reynolds number, averaged over zone length
T	saturation temperature of vapor
T_0	reference temperature of vapor
T_e', T_c'	water inlet temperature in the evaporator and the condenser
T_e'', T_c''	water outlet temperature in the evaporator and the condenser
u	radial vapor velocity
W	molar mass
w	axial vapor velocity
$\overline{W}$	axial vapor velocity, averaged over the cross section
z	axial coordinate
β	momentum factor
δ	dimensionless radial coordinate
ζ	relative pressure recovery
μ	viscosity of vapor
ρ	density of vapor

Subscripts:

1	beginning of evaporator
2	middle of adiabatic zone
3	end of condenser

Introduction

The gasdynamic phenomena of the vapor flow in a heat pipe are rather complex. Radial mass injection in the evaporator due to evaporation leads to an accelerating flow field, whereas, radial mass extraction in the condenser has the opposite effect. In both evaporator and condenser the velocity profiles deviate considerably from developed laminar or turbulent pipe flow. Uncertainties exist especially under conditions of varying vapor densities at high Mach numbers and when inertia effects in the vapor dominate the flow, i.e., generally at $Re_r \gtrsim 1$. Re_r is the radial Reynolds number defined by

$$Re_r = \frac{\rho u D}{2\mu} = \frac{1}{2\pi\mu h_{fg}} \frac{dQ}{dz} \quad . \tag{1}$$

Additionally, a high radial Reynolds number leads to turbulence in cylindrical condensers even if the axial Reynolds number is well below the transition value for pipe flow. Experimental evidence for this phenomenon at axial Reynolds numbers of only a few hundred is given by Quaile and Levy[2] who measured the velocity profiles in a porous tube with suction by means of a hot wire probe. They found that the transition to turbulence first occurs at the downstream part of the condenser. The beginning of this turbulent region moves upstream with increasing radial Reynolds number. At $Re_r = 6$ it extends over 30% of the tube length, at $Re_r = 10$ over 60%. They measured the pressure increase in the condenser up to $Re_r = 16$ and found good agreement with laminar flow predictions. The measurements of the pressure recovery by Quaile and Levy were made at low Mach number and hence the flow was incompressible. Most of the other investigations published so far are limited to even smaller Re_r.[3-6]

The main objective of the present study is, therefore, to investigate pressure recovery in a cylindrical heat pipe at a parameter range of $\overline{Re}_r$ and $\overline{Ma}$ well beyond the investigations published so far. That is at average radial Reynolds number of between 5 and 150. The average Mach number $\overline{Ma}$ ranges between 0.1 and nearly 1.0. For this investigation accurate measurements of the static pressure in the heat pipe as well as steady state operating conditions of the heat pipe at high Mach number are required.

Due to the interaction between the condensate and the counterflowing vapor at high vapor velocities, complicated non-stationary phenomena such as surface waves, flooding in the condenser, surging into the evaporator, and entrainment can occur.[7,8] All these flow phenomena lead to unsteady heat pipe operating conditions and, as a consequence, to variations in the heat pipe power and to significant pressure changes in the heat pipe. Therefore, prior to experimental investigations on pressure recovery, a detailed study on vapor-liquid interaction at high Mach numbers in a cylindrical, gravity-assisted heat pipe was carried out.

The observed flow phenomena are described in the next section. Based on these observations, modifications to the heat pipe were made that suppressed these fluctuations. The basic design features of the heat pipe and the experimental methods are then discussed. The analysis of the experimental data was done with the computer code AGATHE (Analysis of GAsdynamic Transport of HEat) which uses a laminar, two-dimensional flow model and a turbulent, one-dimensional flow model to describe compressible vapor flow in heat pipes.[1] This versatile code is the first one fully adapted for heat pipe analysis for all radial Reynolds numbers Re_r and for Mach numbers up to the velocity of sound. The results of both the experiments and the analysis are also reported.

Liquid-Vapor Interaction

Liquid-vapor interactions in heat pipes have long been a topic of investigation and are of widely shared interest among heat pipe researchers. One of the reasons is the consequences that flow phenomena like flooding, surging and entrainment have on heat pipe operation and specifically the effect on overall heat transfer capability and performance limitations. However, these flow phenomena are not yet fully understood. A successful model to predict entrainment in heat pipes with simple wick structures was developed by Prenger and Kemme.[9] In their experiments performance limits were defined as operating points where the heat pipe is no longer isothermal. But few observations of flow phenomena in cylindrical heat pipes have been made. The observations presented in this paper are the results of a preliminary study on flow phenomena with the objective to obtain information on vapor-liquid interaction at high Mach numbers and at high radial Reynolds numbers.

Figure 1 is a schematic of the heat pipe used for the flow observations. It was made of an aluminum alloy. The overall length was 800 mm and the diameter of the vapor duct was 47 mm. The length of evaporator and condenser were each 200 mm. The wick structure consisted of longitudinal grooves with an aperture angle of 120° and a width of 1 mm. The heat pipe was heated and cooled by two independent external water loops and operated in the vertical, gravity-assisted position. Dodecane, used as the working fluid, has a low heat of vaporization, a low vapor pressure in the operating temperature range and a low sonic velocity. These properties enable the working fluid to attain high Mach numbers as well as high radial Reynolds number at a fairly low heat pipe power and heat input per unit length dQ/dz, respectively. The operating temperature range was between 25 and 55°C and the heat pipe power was varied between 15 and 450 W. A window at the downstream end of the heat pipe allowed direct observation of the condensate flow characteristics. Thermocouples were located in the vapor duct at the beginning, the middle, and the end of the heat pipe. Power transport by the heat pipe was measured at the calorimeter using coolant flow rate and temperature rise measurements.

At small Mach number the heat pipe operated smoothly, i.e., no unstable condensate flow was observed. No temperature fluctuations could be detected and the heat pipe power was fairly constant. Only capillary flow was observed, i.e., the condensate remained within the grooves. However, the evaporator was not uniformly wetted and exhibited large dry zones.

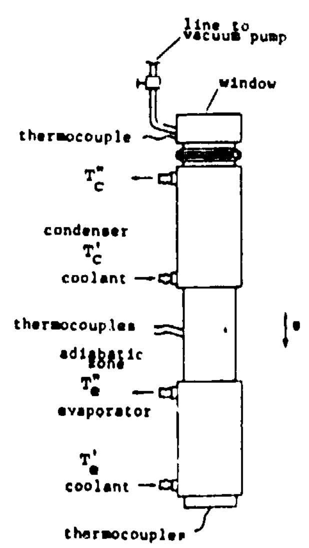


Fig. 1. Schematic of the vertical gravity-assisted heat pipe.

At increasing Mach number, the grooves at the beginning of the condenser filled up and free flow of the condensate occurred. Some small wave formations were also observed. At sufficiently high vapor velocities the unsteady flow behavior became much more pronounced. Due to the high dynamic forces of the vapor flow the condensate accumulated at the beginning of the condenser and a thick liquid ring was observed (Fig. 2). The reduced condensate return flow caused a receding liquid front in the evaporator, which was then partially dried out. With the decrease of evaporation the vapor flow decreased and the dynamic pressure on the condensate ring diminished. As a consequence, the hydrostatic pressure predominated again and the condensate surged into the partly dry evaporator. This was followed by rapid evaporation and the process was repeated. This periodic process caused unstable heat pipe operation and fluctuating heat pipe power. Considerable temperature variation at each thermocouple location in the heat pipe was detected. The cycle duration of this periodic process was approximately 2 seconds. Similar processes have been observed by Busse and Weber[10] in an annular, gravity-assisted heat pipe.

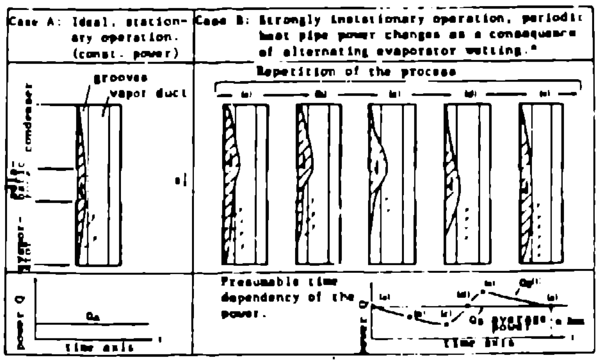


Fig. 2. Qualitative plot of two basic condensate flow phenomena in the vertically operated heat pipe with uniform longitudinal V-grooves (simplified, one-sided graph). Case B shows five instantaneous images of the periodic process.

* Average power and Mach number higher than in Case A.

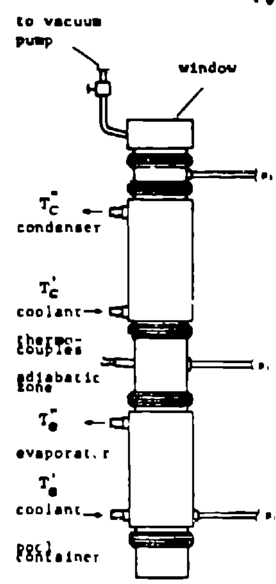


Fig. 3. Schematic of the modular heat pipe.

Pressure Recovery Experiments

A steady internal flow in the heat pipe is required to determine the pressure recovery. It is necessary, therefore, to suppress the fluctuations of the condensate and heat pipe power described above. This was achieved by developing a modular heat pipe (Fig. 3) with a non-uniform wick structure that included a device for condensate distribution in the evaporator as shown in Fig. 4. This heat pipe consisted of four parts that were joined with flanges and O-ring seals. The heat pipe size was similar to the heat pipe that was used for the flow observations. Except for the adiabatic section the capillary structure was also the same. The adiabatic section, which was made of stainless steel, contained a thick screen wick. A thin-walled ring covered both the screen and the grooves forming a transition section between the adiabatic zone and the evaporator.

Any surge of liquid entering the thick screen wick of the adiabatic section was damped. The condensate accumulated in a small circumferential pool between the ring and the wall of the adiabatic section. This intermediate pool supplied each separate flow channel of the evaporator with approximately the same amount of condensate. Using the window at the downstream end of the heat pipe, uniform wetting of the evaporator was confirmed even at high Mach numbers.

The static pressure along the heat pipe was measured by means of a precision capacitance pressure meter at three locations; at the beginning of the evaporator, in the middle of the adiabatic zone and at the end of the condenser. To prevent condensation of vapor in the connecting lines of the pressure meter, tubes and valves were heated. Each series of measurements was made at constant operating pressure, defined as the static pressure in the middle of the adiabatic zone. The heat pipe power and operating temperature varied. The range of radial Reynolds numbers obtained was between 5 and 150 and the average Mach number varied between zero and one. The vapor temperature changed between 25 and 55°C. Figure 5 shows a view of the experimental set-up.

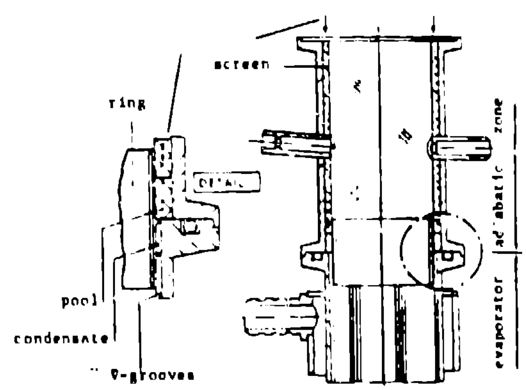


Fig. 4. Design of the adiabatic zone and transition to the evaporator.

The relative pressure recovery is defined as the ratio of the pressure recovery to the total pressure drop or

$$\zeta = (p_3 - p_2)/(p_1 - p_2) \quad . \qquad (2)$$

The saturation temperature T at each measurement location was calculated from

$$p = p_0 \, e^{-Wh_{fg}(1/T - 1/T_0)/R} \quad . \qquad (3)$$

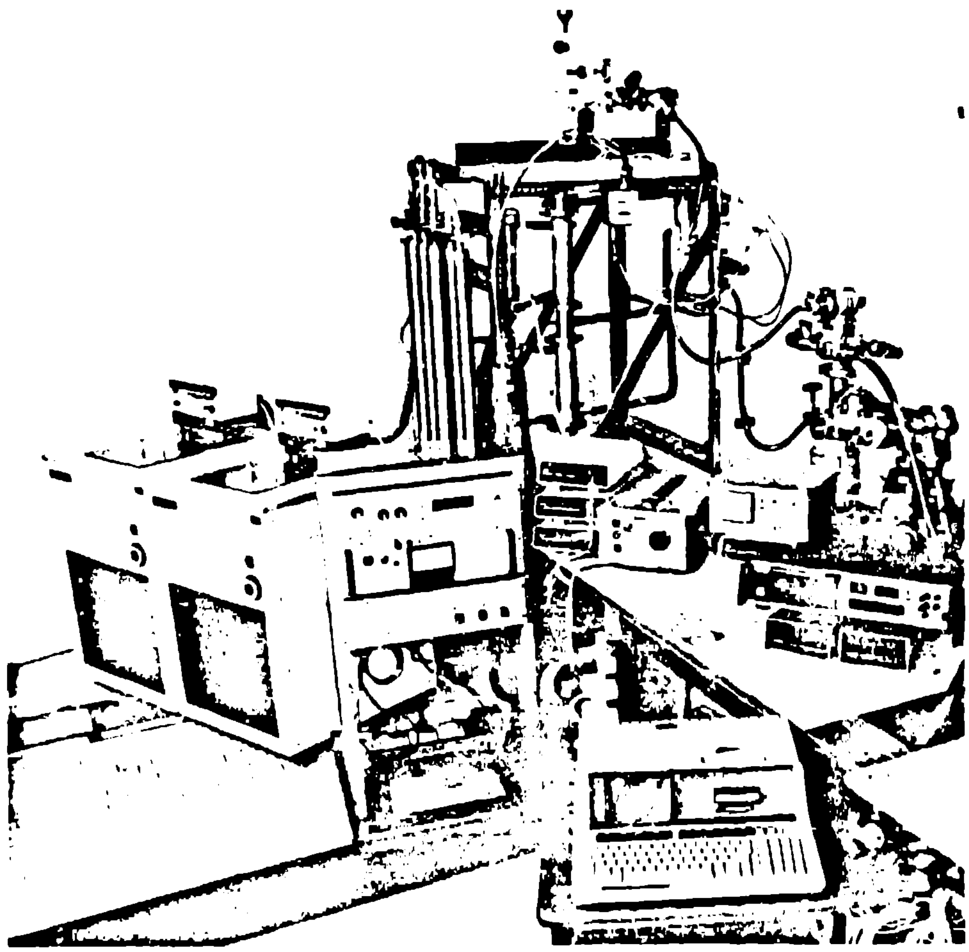


Fig. 5. View of the experimental set-up.

The heat pipe power was determined by an energy balance in the liquid loops using water inlet and outlet temperature and the coolant flow rate.

$$Q = c_p F \left(T_e' - T_e''\right) \tag{4}$$

Nearly uniform radial Reynolds numbers could be obtained by adjusting the coolant water flow rates to match the coolant temperature rise to the saturation temperature rise. The average radial Reynolds number is

$$\bar{Re}_r = Q/\left(2\pi\mu h_{fg} L\right) \quad . \tag{5}$$

The average Mach number in the middle of the adiabatic zone is

$$\bar{Ma}_2 = Q/\left(\frac{pW}{RT} A\, h_{fg} c\right) \tag{6}$$

Analysis

The computer Code AGATHE[1] was developed to evaluate heat pipes operating with compressible vapor flow at Mach numbers up to 1 and at all radial Reynolds numbers. For each heat pipe zone a laminar, two-dimensional calculation or a turbulent one-dimensional calculation with input of empirical factors for describing the turbulence can be chosen. The model features and solution procedures of this code can be found in Ref. 1. Measurements were used to verify laminar theory and to provide empirical data for turbulence.

Which model (laminar, turbulent) to select for each zone can partly be determined by using literature data.[2,11] Injection or evaporation in cylindrical pipes leads to a stabilization of the laminar flow. The rate of injection per unit length is described by the radial Reynolds number. If the radial Reynolds number is high enough, turbulence does not occur even at axial Reynolds number well beyond the transition value for pipe flow which is 2300. Huesman and Eckert[11] found in experiments with a porous tube with injection that at $Re_r = 70$, a transition to turbulence occurred only if the axial Reynolds number exceeded 10,000 at the exit of the tube. The ratio of radial to axial Reynolds number in this case was 0.007.

In the heat pipe used for the present study the maximum axial Reynolds number was < 5200 whereas the corresponding average radial Reynolds number was nearly 150. The ratio Re_r/Re was 0.0288 which is more than four times higher than in the experiments by Huesman and Eckert. This increases the stability of the laminar flow. Therefore, to calculate the pressure variation along the evaporator the laminar model was chosen.

The adjacent adiabatic zone had a length-to-diameter ratio of approximately 2.5, which was very small. A laminar flow entering this zone was not likely to undergo transition to turbulence even at axial Reynolds numbers between 2300 and 5200. Additionally, the velocity profile at the transition from the evaporator to the adiabatic section is a rather flat and stable cosine-profile. Therefore, to describe the flow in this zone the laminar model was also chosen.

Suction or condensation leads to turbulence in cylindrical condensers even if the axial Reynolds number is well below the transition value for pipe flow. Even at very small radial Reynolds numbers $Re_r \geq 4$ Quaile and Levy could detect backflow in the velocity profile close to the wall. Such velocity profiles are unstable. At slightly higher Re_r the onset of turbulence in the flow in the downstream part of the condenser was observed. Generally it can be concluded that the higher the radial Reynolds number the further the beginning of the turbulent region moves upstream. The question is how to describe the flow which is in a state of transition from a laminar to turbulent.

For this purpose two attempts were made to calculate the condenser flow;

a) a laminar, two-dimensional calculation,

b) a turbulent, one-dimensional calculation using an empirical friction factor.

The boundary conditions for the turbulent attempt were based on two assumptions;

1. The boundary layer at the wall will turn turbulent and will spread towards the centerline with increase in the axial coordinate, but the velocity profile along the relatively short condenser will not vary substantially. Therefore the momentum factor β calculated from the laminar velocity profile at the transition from the adiabatic zone to the condenser inlet can be used. The momentum factor β is defined as

$$\beta = \int_0^1 \left(\frac{w}{\bar{w}}\right)^2 d\sigma \quad . \tag{7}$$

2. Suction, or in this case condensation at the wall, prevents the center flow from seeing the state of the wall. Therefore, the friction factor should be independent of the wall roughness. The friction factor for a turbulent flow for this case can be described by Blasius' law describing flow in a smooth tube

$$f = \frac{0.078}{Re^{1/4}} \quad . \tag{8}$$

The iteration procedure for this calculation is described in Ref. 12.

Results

Figures 6 - 9 show the relative pressure recovery for four different operating pressures, as a function of the average radial Reynolds number. Also, the related axial Reynolds number is shown on the same axis. The maximum local Mach number, which occurs in the centerline at the outlet of the adiabatic section, is indicated by arrows for two cases $Ma_{max} = 0.5$ and 1.0. This maximum local Mach number has been determined from the laminar velocity profile which was calculated using AGATHE.

All four curves show nearly the same behavior, a strong increase of the relative pressure recovery with an increase in the radial Reynolds number. It reaches a maximum followed by a decrease beyond $Ma_{max} \geq 1$. In Fig. 9, which shows the data at the highest operating pressure, this maximum is very flat: that is the pressure recovery remains constant over a wide range of radial Reynolds numbers.

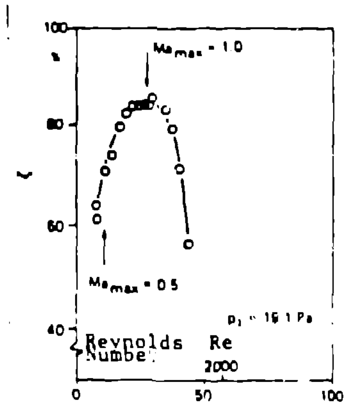


Fig. 6

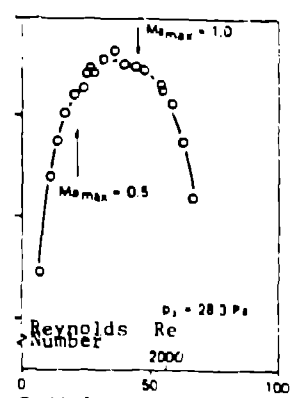


Fig. 7

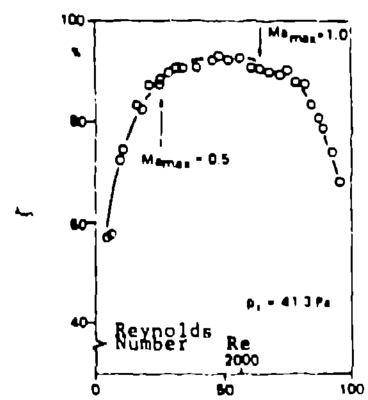
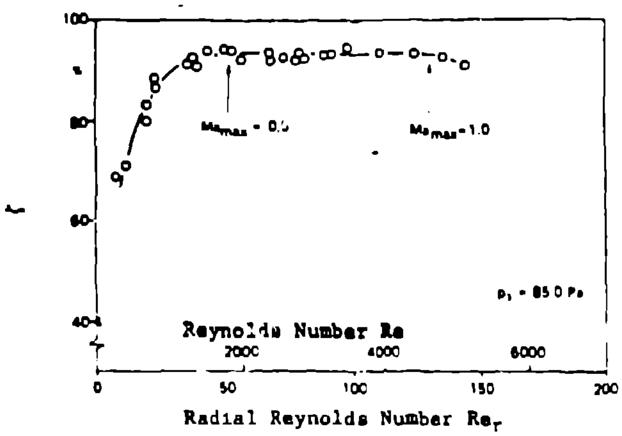


Fig. 8



Figs. 6-9: Pressure recovery $(p_3 - p_2)/(p_1 - p_2)$ in percent versus the radial Reynolds Number for different operating pressures p_2. As a parameter the maximum local Mach number at the end of the adiabatic zone is indicated.

The increase of the pressure recovery with the radial Reynolds number, which is the ratio of inertia to viscous forces, is attributed to a decreasing influence of viscous forces. The decrease of the pressure recovery at maximum local Mach numbers greater than one indicates an additional energy dissipation at supersonic core flow. Apparently if local Mach numbers are close to or greater than one and subsequent condensation occurs, an additional dissipation mechanism exists. An explanation would be that an increasing part of the core flow becomes supersonic and compression shocks might occur in the downstream part of the condenser. The higher the operating pressure, the higher are the radial Reynolds numbers at which supersonic flow occurs. For the operating pressure of 85 Pa, shown in Fig. 9, this radial Reynolds number is about 130.

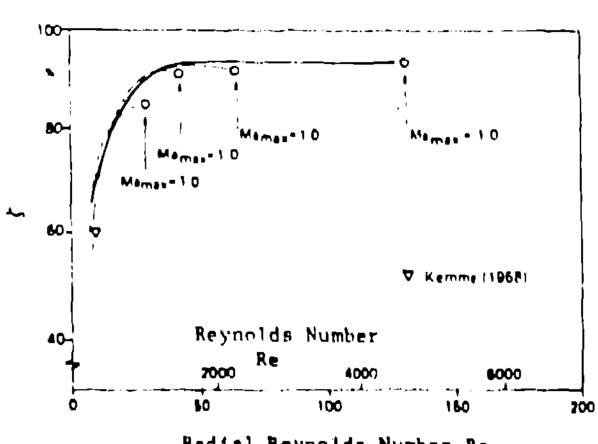


Fig. 10 Comparison of relative pressure recoveries at different operating pressures up to maximum local Mach numbers of one.

Figure 10 is a summary of the increasing part of the pressure recovery curves of Figs. 6 to 9. At subsonic flow the four curves nearly lie together. The relative pressure recovery reaches a maximum of around 93 to 94% at a radial Reynolds number of about 50 and remains constant until the maximum local Mach number exceeds 1.

Also indicated in Fig. 10 is a data point measured by Kemme[4] with a cylindrical, sodium heat pipe. It agrees well with the results of the present study.

Figure 11 shows a comparison between the experimentally obtained pressure recovery curve at small Mach numbers and the calculated ones. As explained in detail in the previous section calculations in both the evaporator and the adiabatic zone were made with the laminar model. The adjacent condenser was subsequently calculated with the laminar or the turbulent model. In Fig. 11 the indication of laminar or turbulent refers to the respective model. In both cases the pressure recovery increases with increasing radial Reynolds number as does the experimental curve, which lies between the calculated ones.

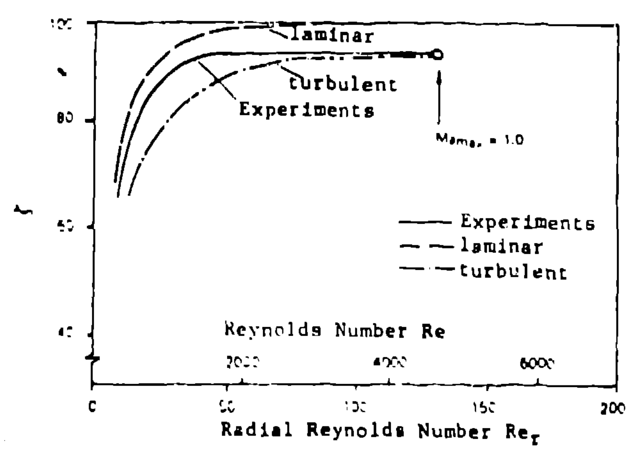


Fig. 11. Comparison of calculated pressure recovery with the experiments.

At small radial Reynolds numbers the laminar model agrees well with the experiments although the condenser flow is in a state of transition from laminar to turbulent flow. This result is also in good agreement with Quaile and Levy's (1973) laminar flow predictions for Re_r up to 16. With increasing Re_r the deviation of both curves become more pronounced. At radial Reynolds numbers above 60, the turbulent model offers the better result. In the interval $80 \leq Re_r \leq 130$ the curve based on the turbulent model shows a very good approach to the experiments. The deviation between both curves is only around 1 - 2%.

These results indicate that at high radial Reynolds number the transition to turbulence may have already occurred at the beginning of the condenser. At smaller Re_r, however, the entrance part of the condenser, where the highest velocities (and therefore the highest energy dissipation) exist, may still be laminar. This could explain the better results with the laminar flow model in this parameter range.

Conclusions

In a cylindrical gravity-assisted heat pipe operating at high Mach numbers unsteady condensate flow as a consequence of the vapor-liquid interaction was observed. The condensate accumulated at the beginning of the condenser in a ring-like pattern. A reduction in condensate return flow caused a partial dry-out of the evaporator. As a consequence the dynamic forces diminished and the condensate surged into the partly dry evaporator. This was followed by rapid evaporation and the process was periodically repeated.

Based on these observations, a modular heat pipe was designed to provide steady-state operation. This provision enabled accurate measurements of the pressure recovery in the heat pipe at Mach numbers up to the velocity of sound and at high radial Reynolds numbers between 5 and 150.

The pressure recovery depends strongly on the radial Reynolds number. It increases with Re_r from a value of about 50% at $Re_r \approx 5$ to a maximum of 93% at a radial Reynolds number of about 50. At $50 \leq Re_r \leq 130$ the pressure recovery remains practically constant at 93%. A strong decrease in pressure recovery occurs independent of Re_r if the local Mach numbers reach or exceed 1.0. The decrease of the pressure recovery at local Mach numbers above one indicates an additional energy dissipation at a supersonic core flow.

Measurements of pressure recovery for subsonic flow at Re_r up to 30 compared well with a laminar two-dimensional model. At $Re_r \geq 80$, however, the turbulent model showed excellent agreement with the measured pressure recovery.

Acknowledgement

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